APPLIANCES FOR THE TRANSMISSION OF POWER

BY

HARRY McCARTHY

THESIS FOR DEGREE OF BACHELOR OF SCIENCE
IN MECHANICAL ENGINEERING

COLLEGE OF ENGINEERING
UNIVERSITY OF ILLINOIS

PRESENTED JUNE 1902

UNIVERSITY OF ILLINOIS

May 29, 1902 190

THIS IS TO CERTIFY THAT THE THESIS PREPARED UNDER MY SUPERVISION BY

Harry McCarthy

ENTITLED Appliances for the Transmission of Power

IS APPROVED BY ME AS FULFILLING THIS PART OF THE REQUIREMENTS FOR THE DEGREE

OF Bachelor of Science in Mechanical Engineering

L. P. Breckenridge

HEAD OF DEPARTMENT OF Mechanical Engineering

PREFACE.

-o-

This Thesis deals only with the appliances for the transmission of power by mechanical gearing, which are in common use in the factories and shops of this part of the United States. Such appliances as are used in other places have not been considered. For instance, in England and in some of the eastern states bevel gears and vertical shafts are used in mills, whereas in this locality belting would be used.

TABLE OF CONTENTS.

APPLIANCES FOR THE TRANSMISSION OF POWER.

-o-

INTRODUCTION.

Power is never available for work at the point of its development, so that in order to utilize power in doing work , some form of power transmission becomes necessary. The oldest method of transmission is by means of mechanical gearing and was the only form used or known until comparatively recent times. The appliances for this transmission were so crude that power could only be transmitted for very short distances and that with great loss. It has not been many years since all power in factories and mills was transmitted by means of cumbersome wooden shafts upon which there were wooden drums for the driving belts when they were used, but more often wooden gears were used to take the power from the line shaft. However, that time is passed and now we can transmit power with comparatively small loss for hundreds of miles by means of new methods and appliances. At present the following methods are employed for power transmission:

(1) Mechanical gearing, transmitting power and rotary motion.

(2) Hydraulic apparatus, conveying power alone.

(3) Steam, conveying heat and power.

(4) Air, conveying power alone.

(5) Gas, conveying light, heat and power.

(6) Electricity, conveying light, heat and power.

The first, that of mechanical gearing, is the only form of transmission that can be used seperately and is by far the most usual method employed. Whenever one of the last five methods is employed the first method is used in combination with it. When power is transmitted by any except the first method it must first be transformed into rotary mothion by means of some form of a motor. Rotary motion can be conveyed only by means of some form of mechanical gearing. For instance, when power is transmitted by electricity from a distance, it is not available for work until it is first transformed into rotary motion by means of an electric motor. This power, now in the form of rotary motion, can only be conveyed from the motor to the point at which work is done by some form of mechanical gearing. If the motor is connected directly to the machine, as is coming to be the case, the power is transmitted by link work, shafting, belts, gears or some of the appliances used in this form of transmission from the motor through the machine to the points at which it is absorbed in doing work and overcoming frictional resistances. This is likewise true of transmission by air, steam, hydraulic apparatus and gas.

For short distances, mechanical gearing is the most economical form of transmission, but when the distances are great, this form is either impossible or so wasteful of power, due to the friction in the parts, that its cost becomes prohibitive to its use. For distances up to a few hundred feet shafting is more economical for transmitting power than is wire rope, but for long distances, it cannot by used, as the intermediate mechanism absorbs an important portion of the power by vibrations and friction, and for a distance of several hundred feet we get at one end of the transmission only a small fraction of the power applied at the other end. Thus, " a shaft one mile long would transmit only about one-half the energy imparted to it; while a shaft two miles long running in well made and lubricated bearings would absorb the entire power in friction; in other words, no amount of power would suffice to turn it."[1]

Power can be more economically transmitted to a considerable distance by means of wire rope than by any other form of mechanical gearing. Tables I and II,[2] taken from "Transmission of Power by wire Ropes" by Albert W. Stahl, gives the comparative cost of transmitting power by means of air, hydraulic apparatus, electricity and wire ropes. These tables are based on the assumptions that the average cost of one horse-power per hour at the operating station is:

in steam engines of more than 50 HP - - - - 2.07¢ per HP per Hr.
" " " " 10 to 50 HP - - - - - - 5.34¢ " "
" " " " less than 10 HP - - - - - 7.71¢ " "

and for water power, 0.2¢ to 0.4¢ and that the interest and depreciation amount to 14% of the capital outlay, and that the efficiencies areas given in the following table:

EFFICIENCIES

DISTANCE OF TRANSMISSION IN FEET.	ELECTRIC	HYDRAULIC	PNEUMATIC	WIRE ROPE
300	.69	.50	.55	.96
1500	.68	.50	.55	.93
3000	.66	.50	.55	.90
15000	.60	.40	.50	.60
30000	.51	.35	.50	.36
60000	.32	.20	.40	.13

---

1. Albert W. Stahl M.E. "Transmission of Power by Wire Ropes." Which is taken fron Beringer's Kritische Verleichung der Electrischen Kraflubertragung.
2. Albert W. Stahl. "Transmission of Power by Wire Ropes."

It will be seen from the table of commercial efficiencies that wire rope is most efficient up to about three miles beyond which electricity and pneumatic transmission is most efficient. However, when we take into account the capital outlay we find that the wire rope is cheapest only to a distance of three quarters of a mile beyond which the electrical transmission is cheapest. It will also be noticed that the hydraulic and pneumatic transmission are never the cheapest as far as cost of transmission is concerned. There are cases, however, where these forms of transmission are preferable to any other.

TABLE I

STEAM-POWER TRANSMITTED.

PRICE IN CENTS OF ONE HORSE-POWER PER HOUR AT RECEIVING STATION.

Maximum Horse-Power Transmitted	Distance of Transmission In Feet	Electric	Hydraulic	Pneumatic	Wire Rope
5	300	4.6	5.1	5.5	2.3
	1,500	4.7	5.8	6.0	2.9
	3,000	4.9	6.4	6.7	3.8
	15,000	5.8	13.2	10.6	11.0
	30,000	6.7	21.3	19.3	21.1
	60,000	10.5	38.5	33.9	46.0
10	300	4.0	4.8	5.1	2.3
	1,500	4.2	5.2	5.4	2.8
	3,000	4.3	5.7	5.8	3.5
	15000	5.1	10.3	9.1	9.1
	30,000	6.3	15.6	12.7	17.2
	60,000	9.8	29.0	21.1	38.7
50	300	3.8	3.3	4.1	2.2
	1500	3.9	3.4	4.3	2.4
	3000	4.0	3.6	4.4	2.6
	15000	4.6	5.9	5.8	5.1
	30000	5.6	8.5	7.2	9.1
	60000	8.6	15.8	10.7	22.5
100	300	3.6	3.3	4.0	2.2
	1,500	3.7	3.4	4.1	2.3
	3,000	3.9	3.6	4.2	2.5
	15,000	4.4	5.8	5.3	4.5
	30,000	5.3	8.4	6.3	7.8
	60,000	8.3	13.9	9.1	19.7

TABLE II

WATER-POWER TRANSMITTED.

PRICE IN CENTS OF ONE HORSE-POWER PER HOUR AT RECEIVING STATION.

Maximum Horse Power Transmitted	Distance of Transmission in Feet	Electric	Hydraulic	Pneumatic	Wire Rope.
5	300	0.71	0.59	0.81	0.22
	1 500	0.73	0.77	0.95	0.38
	3 000	0.75	0.97	1.17	0.61
	15 000	0.89	2.79	2.57	2.53
	30 000	1.05	5.06	4.86	5.06
	60 000	1.70	9.70	9.01	9.84
10	300	0.54	0.50	0.71	0.18
	1 500	0.56	0.61	0.77	0.34
	3 000	0.58	0.75	0.89	0.50
	15 000	0.73	1.92	1.78	1.94
	30 000	0.95	3.12	2.88	3.87
	60 000	1.44	6.42	8.04	8.10
50	300	0.46	0.30	0.44	0.18
	1 500	0.48	0.36	0.48	0.22
	3 000	0.52	0.44	0.56	0.26
	15 000	0.58	0.93	0.89	0.77
	30 000	0.62	1.54	1.31	1.46
	60 000	1.11	2.89	2.19	3.26
100	300	0.40	0.32	0.44	0.16
	1 500	0.44	0.34	0.46	0.20
	3 000	0.46	0.38	0.48	0.22
	15 000	0.52	0.87	0.73	0.56
	30 000	0.65	1.46	0.97	0.79
	60 000	1.01	2.31	1.68	2.40

See page 3.

BELT GEARING

Belt gearing is the appliance of mechanical gearing for power transmission which employs a flexible connecting piece termed a belt or strap, to drive a rotating piece called a pulley, by means of its frictional resistance to slipping. The belt always acts by tension. This is the form of transmission usually employed in America to transmit power from the engine or motor to the line shaft and then from the line shaft to the machines or counter shafts. In England, gears are used in many of the places where we use belting. Belting has the advantage, however, of being quieter and running with smoother motion than gears, due to the fact that shocks are not readily transmitted through the whole system when belts are used as is done when gears are used.

However, if exact velocity ratios are required, belts are not employed for it is not a positive means of driving. There is almost always a small per cent of slipping of belts on their pulleys. If there is no slipping of the belt on its pulley, the velocity of the belt and the surface velocity of the pulley must be equal. Let v be the velocity of the belt, d_1, d_2 the diameters of the pulleys, and N_1, N_2 their revolutions per minute, then

$$\left.\begin{array}{l}\pi d_1 N_1 = v \\ \pi d_2 N_2 = v\end{array}\right\} \therefore \frac{d_1}{d_2} = \frac{N_2}{N_1}$$

the revolutions per minute vary inversely as the diameters of the pulleys. This is only approximately true, however, in actual problems for belts have some thickness and the equations assume that the belt is infinitely thin. If t represent the thickness of the belt, then the effective diameters of the pulleys are $d_1 + t$ and $d_2 + t$. Then

$$\frac{N_2}{N_1} = \frac{d_1 + t}{d_2 + t}$$

The thickness of the belt however, is usually small as compared with the diameter of the pulley so that t may usually be neglected without causing any important error, but it should be noted that the effective radius of a pulley is the distance from the center of the belt to the center of the pulley.

Materials of Belting.

Various substances have been tried for belts, such as gutta-percha, steel, intestines of animals, cotton, and leather of every kind and tanned in nearly every way. The only belts that have been successful, however, have been leather and cotton and are the only belts that need be considered. The term cotton belts includes those coated with India rubber and usually called gum belts or rubber belts.

There is a variety of different kinds of leather belts which have been used, but the best leather belts are made from ox hide tanned with oak bark. The best green hides are selected for belts and then after the tanning process is completed, which for good leather belts lasts from six to eight months, the belting is cut from the back of the hide called the butts, as shown in Figure 1. Strips are cut from the portion of the hide a, b, c, d, and cemented or riveted together in order to get the required length. The larger sizes of belts are built up of two or three thicknesses of leather. They are fastened together by cement or sometimes by cement and rivets. A double or triple belt has the advantage of being more uniform in strength than single belts. The strength of strips cut from the same hide will often vary as much as 47%.

Leather belts, on account of their greater durability, are most suitable for belts which are subject to rubbing action, to animal oils or are shifted from one pulley to another. They are not, according to

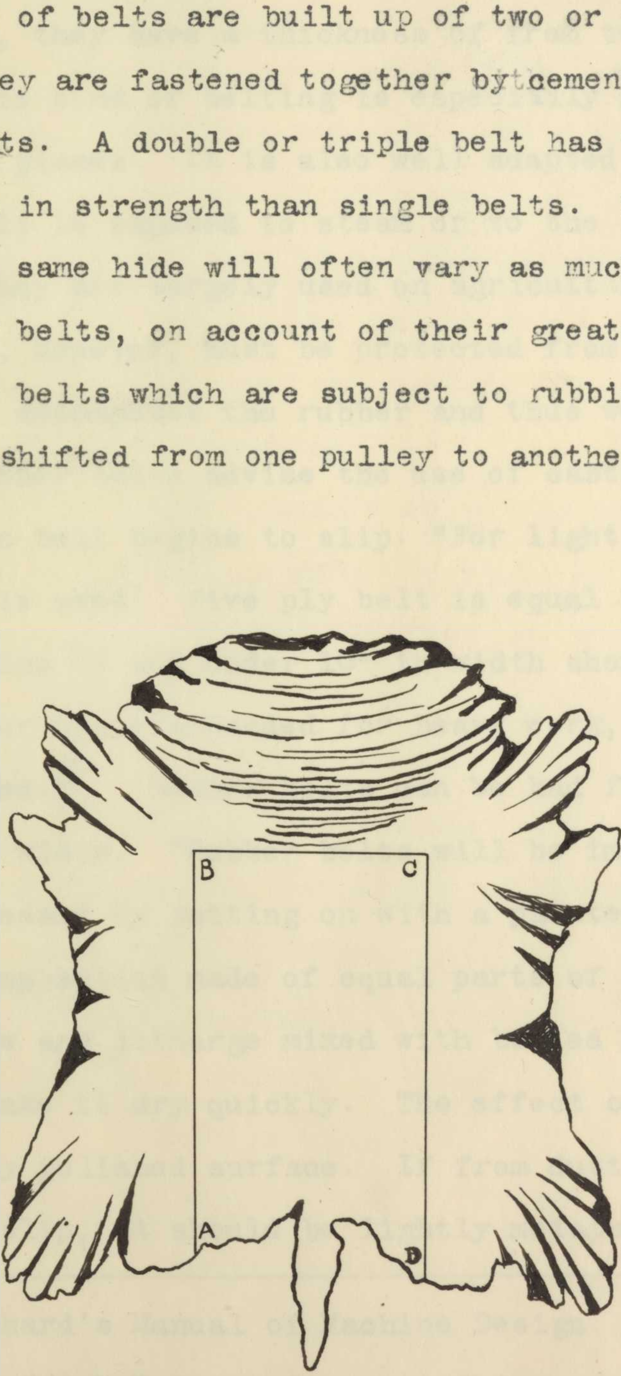


Figure 1.

cording to Richards[3], so suitable for main driving belts as are gum bands of good quality. Notwithstanding, leather is more commonly used.

Gum belts, or rubber belts, are made of layers of heavy canvass and vulcanized rubber. They can be obtained from two to eleven ply; that is, they have a thickness of from two to eleven layers of canvass. This kind of belting is especially adapted for use in damp and hot places. It is also well adapted for use in places where the belt is exposed to steam or to the action of a damp atmosphere. They are largely used on agricultural machinery. This kind of belt, however, must be protected from grease and animal oils as this decomposes the rubber and thus weakens the belt. The makers of rubber belts advise the use of castor oil in small quantities if the belt begins to slip. "For light service three-ply rubber belt is used. Five ply belt is equal in strength to single leather. Belts 6" and under 10" in width should be 7 ply. For large belts or those intended for heavy work, 9 ply and 11 ply should be used."[4] Rubber belts can be had from one inch to sixty inches in width. "Rubber belts will be improved, and their durability increased by putting on with a painters brush, and letting it dry, a composition made of equal parts of red lead, black lead, French yellow and litharge mixed with boiled linseed-oil and japan enough to make it dry quickly. The effect of this will be to produce a finely polished surface. If from dust or other cause, the belt should slip, it should be lightly moistened on the side next

---

3. See Richard's Manual of Machine Design. pp. 61.

4. Catalog Jewel Belting Company, Chicago.

the pulley with boiled linseed oil."[5] Rubber belts have greater adhesion than leather belts and are therefore less liable to slip.

Width of Belts.

There is probably no problem in mechanical engineering upon which authorities disagree as much as the width of belts or what is the same thing, the horse power which a belt will transmit. Nearly every writer on the subject deduces or else gives formulae for the width of a belt and no two of them are alike. The size belt obtained for the transmission of a given horsepower by one formula is just one-half the size obtained by another formula, or in other words, there is a variation of over one hundred per cent. in the results as given by the different formulae.

Evidently the power transmitted by the belt will be equal to the work done in one minute which is equal to the pull upon the belt times the velocity of the belt in feet per minute.

Let T_1 = tension on tight side

T_2 = " " slack side

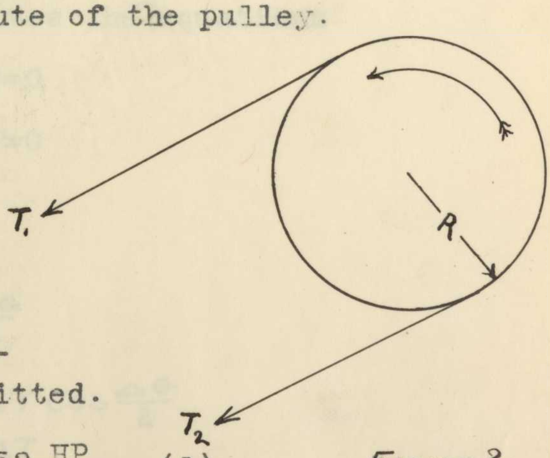
R = radius of pulley in feet

N = revolutions per minute of the pulley.

$T_1 - T_2$ is the effective pull on the pulley.

$(T_1 - T_2)2\pi RN$ = Ft. pounds work per min.

$\therefore$ H.P. $= \dfrac{(T_1 - T_2)2\pi RN}{33,000}$ = Horse-power transmitted.



Transposing, we have $T_1 - T_2 = 5,252 \dfrac{HP}{RN}$ (1)

Figure 2.

5. Kent's Mechanical Engineers' Pocket Book. pp. 887.

The following method for the determination of T_1 and T_2 is taken from Professor Goodenough's notes on Mechanics of Machinery.[6]

Let θ = arc of contact expressed in radians

μ = coefficient of friction between belt and pulley

ε = base of Naperian system of logrithms = 2.7182.

T_2 = tension on slack side

T_1 = " " tight side.

In Figure 3 a short portion of Belt has an arc of contact subtending the angle $\Delta\theta$ at the center of the pulley. Let the tension at one end be T and that at the other end be $T+\Delta T$; evidently each of these tensions make an angle of $\frac{\Delta\theta}{2}$ with the center line MN. The pressure between the

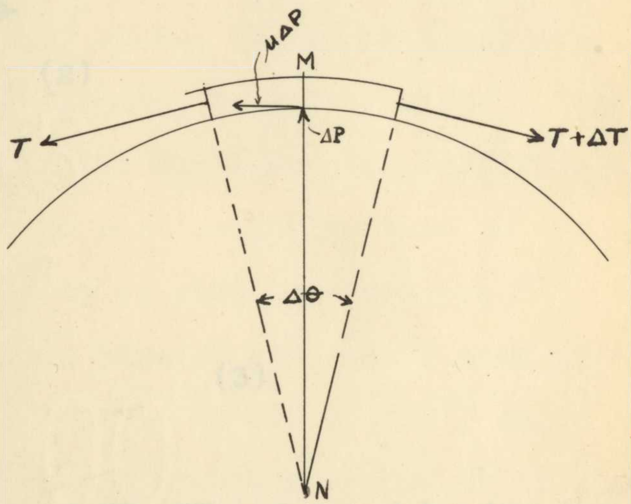


Figure 3.

the portion of belt and the rim is ΔP; and the friction between them is $\mu\Delta P$. The piece of belt is held in equilibrium by the forces T_1, $T+\Delta T_1$, ΔP and $\mu\Delta P$. The summation of the horizontal and vertical forces respectively gives the equations

$$T\cos\frac{\Delta\theta}{2} - (T+\Delta T\cos\frac{\Delta\theta}{2}+\mu\Delta P)=0$$

$$T\sin\frac{\Delta\theta}{2} + (T+\Delta T)\sin\frac{\Delta\theta}{2} - \Delta P=0$$

Simplifying,

$$\mu\Delta P = \Delta T\cos\frac{\Delta\theta}{2}$$

$$\Delta P = (2T+\Delta T)\sin\frac{\Delta\theta}{2}$$

and eliminating ΔP,

$$\mu(2T+\Delta T)\sin\frac{\Delta\theta}{2} = \Delta T\cos\frac{\Delta\theta}{2}$$

$$\text{or}\quad \mu(2T+\Delta T)\tan\frac{\Delta\theta}{2} = \Delta T$$

---

6. See also Unwin "Elements of Machine Design" pp. 376.

Divide both members by $\frac{\Delta\theta}{2}$; thus,

$$\mu(2T+\Delta T)\frac{\frac{\tan\Delta\theta}{2}}{\frac{\Delta\theta}{2}} = 2\frac{\Delta T}{\Delta\theta}$$

$$\text{limit}\left[\mu(2T+\Delta T)\frac{\frac{\tan\Delta\theta}{2}}{\frac{\Delta\theta}{2}}\right] = \mu\ \text{limit}\ (2T+\Delta T)\ \text{limit}\left(\frac{\frac{\tan\Delta\theta}{2}}{\frac{\Delta\theta}{2}}\right) = 2\mu T;$$

hence, since $\text{limit}\ \frac{\Delta T}{\Delta\theta} = \frac{dT}{d\theta}$, $\frac{dT}{d\theta} = \mu T$.

or $\frac{dT}{T} = \mu d\theta$

Integrating, $\int_{T_2}^{T_1}\frac{dT}{T} = \log_\varepsilon\frac{T_1}{T_2} = \mu\theta$

$$\therefore \frac{T_1}{T_2} = \varepsilon^{\mu\theta} \qquad (2)$$

Let $n = \frac{T_1}{T_2} = \varepsilon^{\mu\theta}$

Then, since $T_1 - T_2 = 5{,}252\frac{HP}{RN}$

$$T_1 = 5{,}252\frac{HP}{RN}\frac{n}{n-1}$$

$$(3)$$

$$T_2 = 5{,}252\frac{HP}{RN}\frac{1}{n-1}$$

Having T_1 and T_2 expressed in terms of HP, RN and n and having the value of n, the breadth of the belt can be determined as follows:

Let b = breadth of belt in inches

t = thickness of belt in inches

f = working stress per square inch.

Now, $T_1 = fbt$

and from equation (2)

$$T_1 = T_2\varepsilon^{\mu\theta}$$

$$T_2 = \frac{T_1}{\varepsilon^{\mu\theta}}$$

then $T_1 - T_2 \quad fbt - \frac{btf}{\,} \quad fbt\left(1 - \frac{1}{\varepsilon^{\mu\theta}}\right)$

Substituting this value of $T_1 - T_2$ in equation (1) we have

$$fbt\left[1 - \frac{1}{\varepsilon^{\mu\theta}}\right] = 5{,}252\frac{H.P.}{PN};$$

therefore, $$b = \frac{5252\frac{HP}{RN}}{ft\left[1 - \frac{1}{\varepsilon^{\mu\theta}}\right]} \qquad (4)$$

From the equation (4) it will be seen that the width of a belt to transmit a given horsepower will depend upon the following factors: the arc of contact θ, the coefficient of friction μ, the wor working stress of the belt and the product of R and N which is proportional to the velocity in feet per minute, and also upon the thickness of the belt.

The coefficient of friction μ, as determined by Briggs and Towne, vis, 42% has been used for many years, but as this was determined with a slip of 200 feet per minute, it is too high. "Professor Lanza's numerous tests made at the Massachusetts Institute of Technology indicate that 27% is the most suitable value to use for a low rate of slip of leather belts on cast iron pulleys."

Flather[7] assumes that the working tension per square inch of a single leather belt with cemented joints to be 350 pounds , while Taylor[8] allows only 150 pounds per square inch for double belts. For laced joints, three-fourths of this value may be used; i.e.265 pounds per square inch.

In average practice t = 0.2 for single belting, but varies from0.16 to 0.25; double belting is supposed to be twice this thickness and varies in the same way.

Substituting these values for t, f, μ, and let θ be $180^{\circ} = \pi$ we get for single belts with cemented joints

$$b = 131 \frac{HP}{RN}$$

or, expressing it in velocity in feet per minute of belt and horsepower, we get

$$b = 825 \frac{HP}{V}$$

7. J. J. Flather,"Engineers' Year Book. Uni. of Minn. 1901. pp 71.

8. "Notes on Belting" F. W. Taylor A. S., M.E. Vol. 15, pp 204.'93

Flather uses $b = 800\frac{HP}{V}$.

For single laced belts

$$b = 175\ \frac{HP}{RN}$$

or, $$b = 1100\ \frac{H.P.}{V}$$

Using Flather's constant of 800, then

$$b = 1050\ \frac{H.P.}{V}$$

Calling the coefficients C in the first form of formula and K in the second, we have in general,

$$b = C\ \frac{HP}{RN}$$

and

$$b = K\ \frac{HP}{V}$$

in which C has values as given in Table III, and K has values as given in Table IV.

There is a loss in bending double and triple belts around the pulleys, unless large ones are used, in that the arc of contact is lessened. When the diameter of the pulley is less than one hundred times the thickness of the belt, some allowance for this loss should be made. The ratio of 10 to 7 is customarily used as a multiplier for all double belts irrespective of pulley diameters, but it is hardly necessary except for pulleys less than eight or nine inches in diameter. This should vary however with the size of the pulleys as given in table V. Calling this multiplier C', we have the values given in Table V.

Table III was determined with an assumed arc of contact of 180°. The values of the coefficients should be multiplied by another coefficient when the least arc of contact on either pulley differs from 180°. The coefficient C" has been derived from equation (2) for different angles of contact from 120° to 240° and is

TABLE III

VALUES OF $C = \frac{RNb}{H.P.}$

KIND OF BELT	KIND OF JOINT	
	CEMENTED	LACED
SINGLE	800	1050
DOUBLE	$\frac{800}{2}$	$\frac{1050}{2}$
TRIPLE	$\frac{800}{3}$	$\frac{1050}{3}$

See page 15

TABLE IV

VALUES OF $K = \frac{Vb}{H.P.}$

KIND OF BELT	KIND OF JOINT	
	CEMENTED	LACED
SINGLE	131	175
DOUBLE	$\frac{131}{2}$	$\frac{175}{2}$
TRIPLE	$\frac{131}{3}$	$\frac{175}{3}$

See page 15

TABLE V.

VALUES OF C' = MULTIPLIER WHEN DOUBLE OR TRIPLE BEELTS ARE USED WITH SMALL PULLEYS

FOR DOUBLE BELTS WHEN DIAM. OF PULLEY =	OR TRIPLE BELTS WHEN DIAM. OF PULLEY =	VALUE OF COEFFICIENT C'
8 INCHES	18 INCHES	10 ÷ 7 = 1.42
12 "	26 "	10 ÷ 8 = 1.25
20 "	42 "	10 ÷ 9 = 1.11

See page 15

TABLE VI.

VALUES OF C" FOR ARCS OF CONTACT.

VALUE OF ANGLE IN DEGREES	IN RADIANS	VALUF OF CO-EFFICIENT C"
120	2.09	1.33
130	2.26	1.27
140	2.45	1.21
150	2.62	1.15
160	2.79	1.10
170	2.96	1.05
180	3.14	1.00
190	3.32	.95
200	3.49	.91
210	3.66	.87
220	3.84	.83
230	4.02	.79
240	4.18	.75
250	4.33	.71

See page 18

given in Table VI.

Thus far the effect of centrifugal force, Fo, set up in the belt has been neglected, but for speed over 2,500 feet per minute it is appreciable and should be considered. "Probably the simplest way to effect this in determining the width of belt is to diminish the allowable working stress f, per square inch of section, subtracting Fo from f (since T must otherwise be increased by this force) that is, the effective tension per square inch cross section is f - Fo. Since we have used f in determining the values of C, it is evident that these values should be multiplied by a factor k_o which is greater than K in the ratio: $f : (f - F_o)$.

Since the weight of a unit element of belt is constant, Fo will be constant for a given speed; but as f = 350 in the case of cemented belts and only 265 for laced, we shall have two sets of values of k_o as shown in Table VII which has been calculated from the formula

$$k_o = \frac{f}{f - F_o} = \frac{f}{f - 0.41\frac{V^2}{g \cdot 3600}}$$

Multiplying these various coefficients together we obtain the width of the belt

$$b = C\,C'\,C''\,k_o \times \frac{HP}{V}$$

From this formula and the preceding tables, the width of a belt can be obtained for any given case if the data are known."[9] By transposing, the HP transmitted by a belt can be determined.

Example: Required the horse power that the driving belt from the motor in the University of Illinois wood shop basement will transmit. It is a double 8" belt, the motor pulley is 9" in diameter and the

---

9. J. J. Flather. "The Year Book." University of Minnesota. 1901.

TABLE VII

VALUES OF COEFFICIENT K_o = MULTIPLIER WHEN CENTRIFUGAL FORCE IS CONSIDERED

VELOCITY IN FEET PER MINUTE	CEMENTED BELT	LACED BELT
2 500	1.06	1.10
3 000	1.10	1.14
3 500	1.14	1.20
4 000	1.19	1.27
4 500	1.26	1.37
5 000	1.34	1.50
5 500	1.44	1.65
6 000	1.58	1.87

See page 18

pulley on the line shaft is 40" in diameter. The line shaft makes 300 R. P. M. and the distance between centers of the pulley is 10' 10". The belt is laced.

$$b = KC'C''k_o \frac{HP}{V}$$

$$HP = \frac{Vb}{KC'C''k_o}$$

From Table	III	we	obtain	C	$\frac{1050}{2}$
"	"	V	"	"	C'	1.1
"	"	VI	"	"	C"	1.15
"	"	VII	"	"	k_o	1.14

Substituting these values, we get

$$HP = \frac{Vb}{KC'C''k_o} = \frac{3000 \times 8}{\frac{1050}{2} x1.1x1.15x1.14} = 32 \text{ HP}$$

The motor is rated at 20 HP.

Distance Between Pulleys.

The simplest arrangement of belting is always the best. The tight side of a horizontal belt should always when possible be on the bottom of the pulleys as then the sag in the loose side tends to increase the arc of contact and thus increase the adhesion, whereas if the loose side is on the bottom, the sag will tend to lessen the arc of contact and decrease the friction making the belt liable to slip. Figure 4 shows this. There should always be a gentle sag in the belt when it is in motion and this is obtained by placing the pulleys far enough apart. Where narrow belts are to be run over small pulleys, 15 feet is a good average distance between pulleys, the belt having a sag of 1½ to 2 inches.

For large belts working on larger pulleys a distance of 20 to

25 feet does well with a sag of $2\frac{1}{2}$ to 4 inches.

For main driving belts working on large pulleys, the distance should be 25 to 30 feet with a sag of 4 to 5 inches.[10]

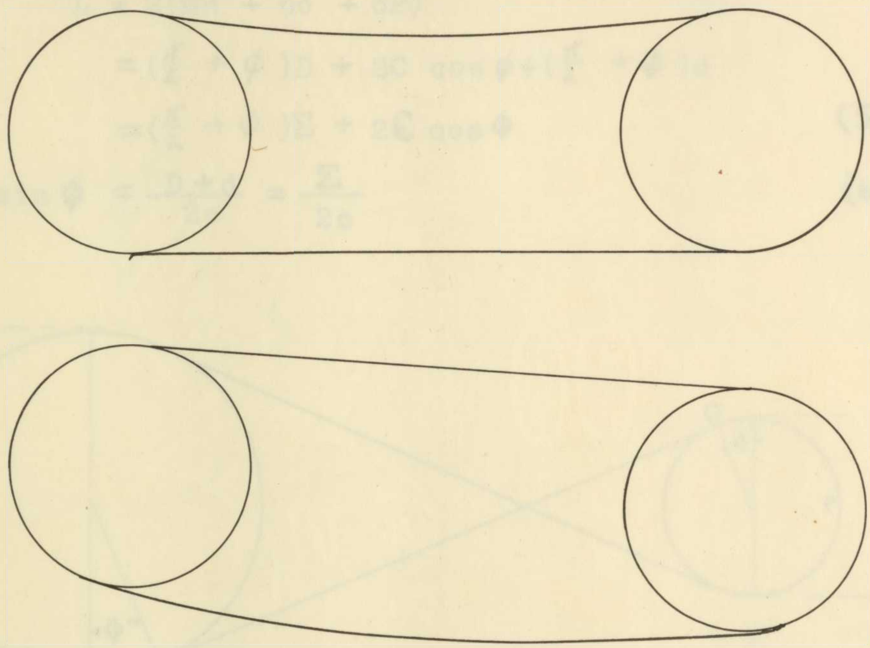


Figure 4.

If there is too great a distance between the pulleys, the weight of the belt will produce a heavy sag, drawing so hard on the shaft as to produce great friction in the hearings and will cause the belt to have an unsteady flapping motion.

Length of Belts.

The length of a belt is found as follows:-

Let D = diameter of large pulley in inches

d = " " small " " "

10. Kent's Engineers' Pocket Book. pp. 885.

c = distance between centers of pulleys in inches

L = length of belt

$D + d = \Sigma$ and $D - d = \Delta$

For a crossed belt (Fig.5.) the length of the belt is,

$$L = 2(mn + no + oP)$$
$$= (\frac{\pi}{2} + \phi)D + 2C\cos\phi + (\frac{\pi}{2} + \phi)d$$
$$= (\frac{\pi}{2} + \phi)\Sigma + 2C\cos\phi \qquad (5)$$
$$\sin\phi = \frac{D+d}{2c} = \frac{\Sigma}{2c} \qquad (6)$$

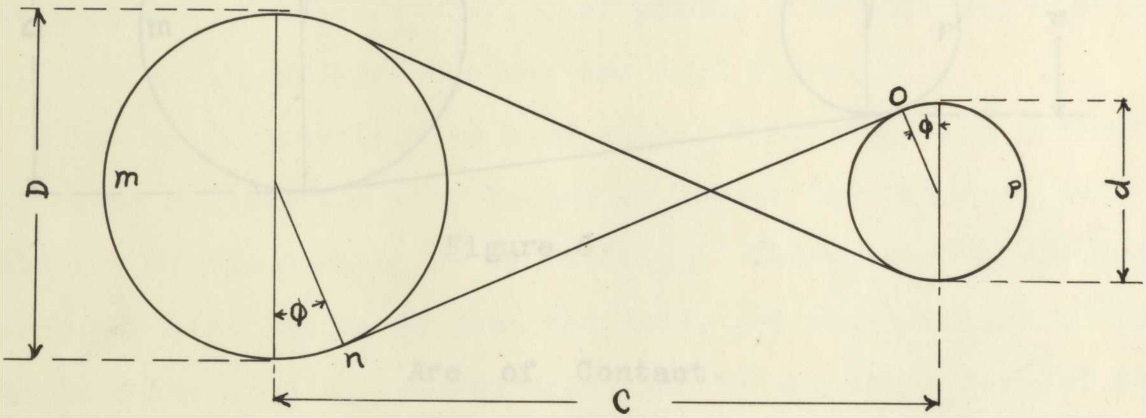


Figure 5

Calculate the value of the sin ϕ and then with a table of sines and cosines find the value of cos ϕ and ϕ. Substitution in equation (5) gives the length. ϕ must be expressed in radians. If expressed in degrees, multiply by 0.0175 to get radians.

If Σ and c are constant for two or more pairs of pulleys the same crossed belt will run on any pair of pulleys of the set because ϕ depends only upon D + d.

When the belt is an open one (Fig.6) the length is,

$$L = 2(mn + no + oP)$$

$$= \left(\frac{\pi}{2} + \phi\right)D + 2c\cos\phi + \left(\frac{\pi}{2} - \phi\right)d.$$

$$\sin\phi = \frac{\Delta}{2c}\ ; \quad \cos\phi\ \sqrt{\left(1 - \frac{\Delta^2}{\Delta c^2}\right)}$$

For an open belt ϕ is generally small, so that $\phi = \sin\phi$, nearly

$$\therefore L = \frac{\pi}{2}\Sigma + 2c\left[\frac{\Delta^2}{4c^2} + \sqrt{\left(1 - \frac{\Delta^2}{4c^2}\right)}\right.$$

$$\frac{\pi}{2}\Sigma + 2c\left[1 + \frac{1}{8}\frac{\Delta^2}{c^2}\right], \text{ nearly.}$$

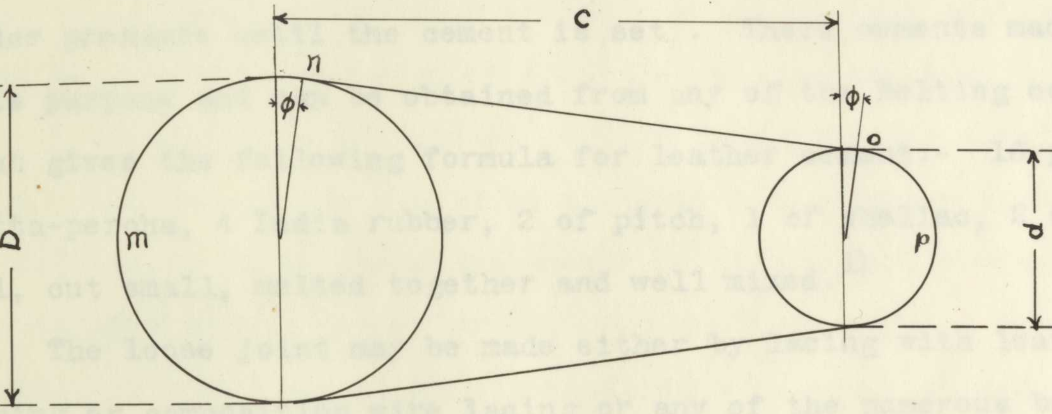


Figure 6.

Arc of Contact.

The arc of contact θ , is always measured on the smaller pulley. It is the arc on the circumference of the pulley, measured in radians, with which the belt is actually in contact. To determine the value of θ, as in the case of determining the length of crossed and open belts, we determine the angle ϕ. Then it will be seen from Figures 5 and 6, that the angle of contact is equal to 180° plus or minus 2ϕ, that is,

$$\theta = 180^\circ \mp 2\phi = \pi \mp 2\phi$$

For crossed belts

$$\theta = 180^\circ + 2\phi = \pi + 2\phi$$

For open belts

$$\theta = 180^\circ - 2\phi = \pi - 2\phi$$

Belt Fastenings.

The joints in a belt should all be cemented except one which should be fastened by some means that would allow of its being taken up when the belt becomes loose due to its stretch. The cemented joints are made by tapering the belt down to a thin edge for about six or seven inches and then cementing the two ends and placing them under pressure until the cement is set . There cements made for this purpose and can be obtained from any of the Belting companies. Kent gives the following formula for leather cement:- 16 parts gutta-percha, 4 India rubber, 2 of pitch, 1 of shellac, 2 of linseed oil, cut small, melted together and well mixed.[11]

The loose joint may be made either by lacing with leather lacing or composition wire lacing or any of the numerous belt hooks that are on the market, or by riveting. There is an advantage in using the wire lacing in that the holes are much smaller in the belt and thus the area is reduced less than in any other form of fastening.

F. W. Taylor[12] recommends the use of all spliced and cemented joints instead of lacing, wiring or using hooks. When belts are subjected to the severest usage the spliced portion should be riveted, iron burrs being used instead of copper. For double belts up to 10 inches in width the splice should be 10 inches long and for belts ten to eighteen inches long the splice should be as long as the belt. When idler pulleys are used on belts he recommends the use of the V splice for double, triple and quadruple belts. For rubber belts the stepped splice should be used and spliced section

---

11. Kent's Mechanical Engineers' Pocket Book. pp 887.

12. Notes on Belting, A.S. M.E. - Vol. 15, pp204. 1893.

should be one or two plies thicker than the rest of the belt.

If lacing is used, the following directions should be observed:[13] "In punching a belt for lacing use an oval punch, the longer diameter of the punch being parallel with the sides of the belt Punch two rows of holes in each end, placed zigzag. In a 3-inch belt there should be four holes in each end, two in each row. In a six-inch belt, seven holes, four in the row. A 10-inch belt should have nine holes. The edge of the holes should not come nearer than 3/4 of an inch from the sides, nor 7/8 inch from the ends of the belt. The second row should be at least 1 3/4 inches from the end. On wide belts these distances should be even a little greater. Begin to lace at the center of the belt and take care to keep the ends exactly in line, and to lace both sides with equal tightness. The lacing should not be crossed on the side next to the pulley. In taking up belts observe the same rules as putting on new ones."

Care of Belts.

There is a great deal said as to which is the proper side of the belt to be run next to the pulley. The fact is that there is very little difference in the results which side is used next to the pulley. When appearance is considered the flesh side of the belt should be run next to the pulley. Those who argue for the other side claim that the flesh side is 30% stronger than the hair side and that, therefore, the hair side should be run next to the pulley where there is the greatest wear.

The motion of driving should run with, and not against the laps of the belt.

---

13. Kent's Mechanical Engineers' Pocket Book. pp 883.

Belts should be kept pliable and soft. To do this they should be cleaned and greased every five or six months. There are a number of belt dressings on the market for the purpose of keeping belts soft, but Tompkins[14] seems to think that none of them are fit to be used. He recommends the following treatment of belts: "Tallow seems to be the only material that is natural to leather, but should never be applied to a belt when dry and covered with dust, for this reason: The solid fats of all animals are composed of three elements vis: sterine, margarine, and oleine.

"Margarine contains a large percentage of margaric acid, which must be kept out of the belt as far as possible. The proper manner to treat a belt when it becomes hard and dry, and to exclude the greater portion of the margarine, is to take it off and lay it upon a clean floor; then, having fastened the ends to the floor to prevent its shrinking, with soap and warm water thoroughly cleanse it and if necessary, scrape it until the surface on both sides is perfectly clean; then prepare some clean tallow by melting it and with a brush apply a thick coat upon the flesh side while it is just soft enough to spread well and while the belt is wet, and then leave it until it becomes perfectly dry. The stearine and margarine are both insoluble in water, and will not enter the pores of the leather while it is wet.

"Margarine has a greater affinity for sterine than it has for oleine, consequently, it remains on the outside and becomes hard before the leather becomes dry enough to absorb it; while the eleine, which has a greater affinity for the leather, seperates from the other two ingredients, and as the water evaporates gradually assumes

---

14. A History of the Plaining Mill. C. R. Tompkins. pp 207

its place leaving the other two on the outside in the form of a white substance much harder than tallow, which may be readily scraped off. Belts treated in this manner about once in six months will be as soft and pliable as new and retain their strength until worn out."

Right Angle Driving.

When two shafts are not parallel and do not intersect they may be connected by a belt provided the pulleys are properly placed. This arrangement is called the half-crossed belts or quarter-twist belts. Unwin says that the single and sufficient condition that the belt may properly run is that the point at which the belt is delivered from each pulley must be in the plane of the other pulley. This condition can be fulfilled only for a belt which always runs in one direction. Figure 7 shows the method of arranging the pulleys for half-crossed belts.

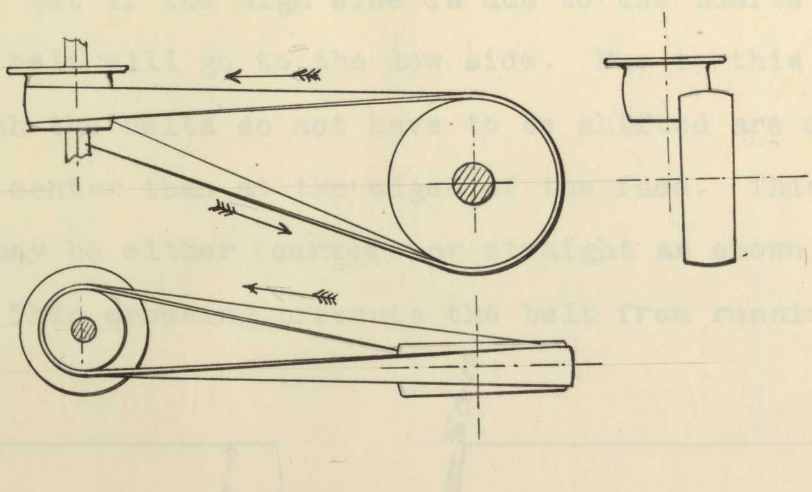


Figure 7.

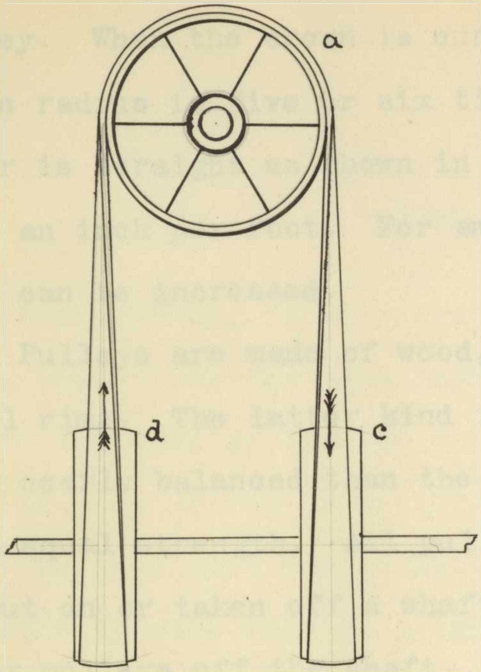


Figure 8.

Another method of driving a shaft which is at right angles to the driving shaft is shown in Fig 8. a is the driving pulley andc is the driven, while d and another just back of a are loose pulleys. With this arrangement the belt will run in either direction and either shaft may be used as the driver. This is preperable to half-crossed belts where the shafts are close together.

Guidance of Belts.

If a pulley is higher on one side than the other, the belt will run to the high side; that is, if the high side is due to the shape of the pulley, but if the high side is due to the shafts not being parallel, the belt will go to the low side. Due to this fact, pulleys upon which the belts do not have to be shifted are usually made larger in the center than at the edges of the face. This is called crowning and may be either curved or straight as shown in Figures 9 and 10. This crowning prevents the belt from running off the

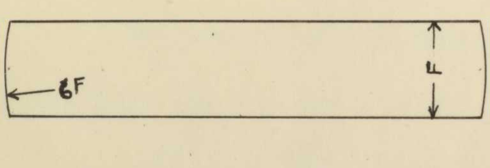


Figure 9.

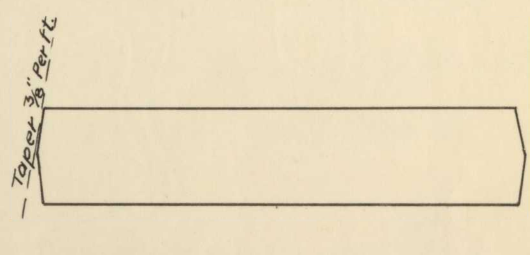


Figure 10 .

pulley. When the crown is curved it should be the arc of a circle whose radius is five or six times the width of the pulley. When the taper is straight as shown in Fig.9, the taper should be from 3/8 to $\frac{1}{2}$ an inch per foot. For small pulleys running at high speed this can be increased.

Pulleys are made of wood, cast iron, steel and cast iron with steel rims. The latter kind is perhaps the best pulley as it is more nearly balanced than the others and can be made much lighter with equal strength. All pulleys should be split so that they may be put on or taken off a shaft without taking the shaft down and all other pulleys off the shaft. The best method of fastening the pulley on a shaft is by means of a set screw and a compression hub.

Large heavy pulleys are keyed to the shaft.

ROPE GEARING.

Rope gearing is very similar to belt gearing, the difference being that ropes are used instead of flat bands and that these ropes run in V-shaped grooves on the face of the pulley. The pulley is used in rope drives is usually called a sheave

The rope used may either be metallic rope or non-metallic. Metallic or wire rope is seldom used for anything but long distance transmission and is called telo-dynamic transmission. The non-metallic ropes are used for comparatively short drives and in many places where leather belting might be used.

Non-metallic ropes are made of hemp, manilla, leather, raw-hide and cotton. The hemp, manilla and cotton ropes are used most for transmission of large amounts of power, but leather ropes or twisted raw-hide ropes are universally used for small amounts of power.

The seperate or English system of rope drive consists of a number of seperate ropes run side by side in grooves in the pulley, whereas in the continuous or American system of rope drive one continuous rope is used and crossed over by means of an idler, which often is used also as a tension pulley.

The same reasoning applies to rope drives as to pulleys only $\frac{T_1}{T_2} = \epsilon^{\frac{\theta\mu}{\sin\beta}}$ where β is one-half the angle that the sides of the grooves make with each other.

Table VIII gives the horse power transmitted by manilla ropes of different sizes running at different speeds and also the smallest size pulleys that can be used with the different size ropes. In

TABLE VIII

HORSE-POWER TRANSMITTED BY ROPES.

WORKING STRAIN = $200 d^2$ #

d = DIAMETER OF ROPE IN INCHES.

VELOCITY OF ROPE IN FEET PER MINUTE	DIAMETER OF ROPE.						
	5/8"	3/4"	1"	1 1/4"	1 1/2"	1 3/4"	2"
1000	1.24	2.25	3.57	5.59	8.02	10.85	14.20
2 000	2.70	3.84	6.84	10.68	15.39	20.93	27.36
2 500	3.30	4.71	8.38	13.10	18.86	25.66	33.54
3 000	3.83	5.46	9.80	15.39	21.87	29.74	38.88
3 500	4.30	6.23	11.09	17.33	24.94	34.07	44.35
4 000	4.74	6.83	12.15	18.98	27.33	37.17	48.59
4 500	5.01	7.24	12.89	20.15	29.00	39.45	51.57
5 000	5.20	7.49	13.29	20.76	29.89	40.65	53.15
5 500	5.29	7.60	13.53	21.14	30.43	41.39	54.11
6 000	5.08	7.32	13.10	20.36	29.32	39.77	52.12
6 500	4.74	6.83	12.13	19.00	27.34	37.21	48.63
7000	4.12	5.93	10.54	16.47	23.72	32.26	42.18
7 500	3.25	4.67	8.32	13.00	18.73	25.42	33.23

TABLE IX

SAG OF THE ROPE BETWEEN PULLEYS

DISTANCE BETWEEN PULLEYS IN FEET.	DRIVING SIDE	SLACK SIDE OF ROPE		
	ALL SPEEDS	4800 FT. PER. MIN.	3600 FT. PER. MIN.	2400 FT. PER. MIN.
40	0 FEET 4 INCHES	0 FEET 7 INCHES	0 FEET 9 INCHES.	0 FEET 11 INCHES
60	0 " 10 "	1 " 5 "	1 " 8 "	1 " 11 "
80	1 " 5 "	2 " 4 "	2 " 10 "	3 " 3 "
100	2 " 0 "	3 " 8 "	4 " 5 "	5 " 2 "
120	2 " 11 "	5 " 3 "	6 " 3 "	7 " 4 "
140	3 " 10 "	7 " 2 "	8 " 9 "	9 " 9 "
160	5 " 1 "	9 " 3 "	11 " 3 "	14 " 0 "

TABLE X.

HORSE-POWER TRANSMITTED BY WIRE-ROPES FROM 3/8"-7/8" DIAM. WITH SIZES AND SPEEDS OF WHEELS.

(ROEBLING)

DIAMETER OF WHEEL IN FEET	NO. OF REVOLUTIONS PER. MIN.	DIAMETER OF ROPE	HORSE-POWER	DIAMETER OF WHEEL IN FEET	NO. OF REVOLUTIONS PER. MIN.	DIAMETER OF ROPE	HORSE-POWER
4	80	3/8	3.3	10	80	11/16	58.4
	100	3/8	4.1		100	11/16	73.
	120	3/8	5.		120	11/16	87.6
	140	3/8	5.8		140	11/16	102.2
5	80	7/16	6.9	11	80	11/16	75.5
	100	7/16	8.6		100	11/16	94.4
	120	7/16	10.3		120	11/16	113.3
	140	7/16	12.1		140	11/16	132.1
6	80	1/2	10.7	12	80	3/4	99.3
	100	1/2	13.4		100	3/4	124.1
	120	1/2	16.1		120	3/4	148.9
	140	1/2	18.7		140	3/4	173.7
7	80	9/16	16.9	13	80	3/4	122.6
	100	9/16	21.1		100	3/4	153.2
	120	9/16	25.3		120	3/4	183.8
8	80	5/8	22.	14	80	7/8	148.
	100	5/8	27.5		100	7/8	185
	120	5/8	33.		120	7/8	222
9	80	5/8	41.5	15	80	7/8	217
	100	5/8	51.9		100	7/8	259
	120	5/8	62.2		120	7/8	300

this, the angle β is taken as being 45° and θ as 180° and μ as .31.(15)

TableIX gives the sag of the rope between the pulleys at various speeds with a maximum working tension of 200# on the rope. This is copied from Kent's Mechanical Engineers' Pocket Book, pp925.

There is a loss of from five to eight percent in rope driving due to the bending of the rope and to the wedging of the rope in the grooves.

Wire Rope Transmission.

For transmissions over 100 feet, wire ropes should be used. For distances over 400 feet intermediate pulleys are used to support the rope, however, in some cases where there is room for the sag, power is transmitted over distances much greater than this without any intervening support. In a transmission at Lockport N.Y. there is a clear span of 1700 feet.

The pulleys for wire rope drive are made of cast iron or steel and for wheels larger than 14 feet in diameter they are made like a bicycle wheel with wrought iron rim and wire spokes. They are made as light as possible to prevent excessive friction. The sides of the grooves make an angle of about 30° with each other. At the bottom of the groove there is ~~there is~~ a trough extending around the pulley. This is filled with filling, as it is called, of some sort, upon which the rope rests. There are various kinds of filling used; wood, rubber, yarn, paper and leather.

Table X[16] gives the horse power transmitted by wire rope and the sizes of pulleys to use.

---

15. J. J. Flather's "Rope Driving" pp 121.

16. Roebling's Table.

SHAFTING.

In factories mills and shops of all kinds where power is used to drive machinery the power is transmitted by long bars of iron or steel called shafts rotating in bearings from which the power is taken off by means of either rope, belt or toothed wheel gearing. Shafting as a means of transmitting power at any considerable distance cannot be used as is shown from the following considerations[17].

Let θ = distortion of shaft in circular measure per unit length

θ° = distortion in degrees;

l = unit length of shaft

L = length of shaft in feet

r = distance of outer fibres from axis $= \frac{d}{2}$

d = diameter of shaft

PR = twisting moment of the shaft

N = number of revolutions of shaft per minute

v = velocity of circumference of shaft   dN

G = modulus of tortion of the material

= two-fiths of the modulus of elasticity

f = maximum torsional stress in outer fibres $\frac{16PR}{\pi d^3}$

W = weight of shaft   3.36# per foot per sq. in. cross sec.

F = load due to friction.

Then

$$\theta = \frac{fl}{Gr} = \frac{32\,PRl}{\pi d^4 G} \qquad (1)$$

17. See J.J.Flather's "Rope Driving" pp 68.

If the angle of torsion is given in degrees, then

$$\theta = \frac{\theta^\circ \times 2\pi}{360} \;;$$

therefore the angular distortion per foot of length will be

$$\theta^\circ = \frac{360}{2\pi} \times \frac{fl \times 12}{Gr} = \frac{36\, f}{\pi G} \times \frac{12\, L}{d} \qquad (2)$$

The working limit of the angle of torsion for steel shafting ought not exceed 0.10 degree per foot in length of the shaft; that is, $\theta^\circ = 0.10\ L$, and for wrought iron shafting $\theta^\circ = 0.075L$.[18] Assuming the shaft to be of steel and substituting the corresponding value for θ° in (2) we obtain

$$0.10\ L \times \pi G d = 360 f \times 12\ L \;;$$

hence, $f = 800d$ if we assume that $G = 11{,}000{,}000$ pounds.

Since the horse-power transmitted by the shaft equals $PR \times \frac{2\pi N}{33000}$, if we substitute the value of PR, $\left(\frac{\pi d^3}{16 f}\right)$, there is obtained

$$H\,P = \frac{\pi d^3}{16} \times f \, \frac{2\pi N}{33000} \;;$$

but the velovity at the circumference of the shaft is $v = \pi dN$, also $f = 800d$; hence,

$$H\,P = 0.0095 d^3 v \qquad (3)$$

If the bearings are well worn and fitted to the shaft the resistance due to friction will probably lie between the limits $\frac{\pi}{2}\phi W$ and $\frac{4}{\pi}\phi W$[19], or between $1.57\phi W$ and $1.28\phi W$, where ϕ is a coefficient, which in the present case we shall assume equal to 0.06.

Taking the lesser value, we shall have $F = \frac{4}{\pi}\phi W$, where F is the force at the circumference of shaft necessary to overcome the journal friction. If there are no pulleys on the shaft, then,

$$W = \frac{\pi}{4} d^2 L \times 3.36;$$

the horse power exerted to over come the friction will then be,

18. Reuleaux; "Der Konstrukteur."

19. Unwin.

$$H\,P_o = \frac{Fv}{33000} = \frac{4}{\pi}\phi \times \frac{\pi}{4}\,\frac{d^2L \times 3.36\,v}{33000} = 0.061\,d^2LV.$$

Expressed as a ratio the percentage of power required to overcome friction will be

$$\frac{HP_o}{HP} = \frac{0.061\,d^2Lv}{0.0095\,d^3v} = \frac{0.061L}{0.0095\,d}$$

$$H\,P_o = 0.00063\frac{L}{d} \times H\,P = \frac{L}{d} \times \frac{HP}{1585} \qquad (4)$$

That is, for a steel shaft whose diameter is one inch the horse power required to overcome the friction in a length of 1585 feet will be equal to the total allowable transmitting capacity of the shaft.

For wrought iron shaft, $\theta^\circ = .075L$ and $HP_o = .0075\,d^2v$ from which may be determined the value of the ratio

$$\frac{HP_o}{HP} = 0.0008\,\frac{L}{d}, \text{ or, } HP = \frac{L}{d} \times \frac{HP}{1250}$$

Table XI gives limit of length of steel shafting calculated from these formula and Table XII gives the limit of length for wrought iron shafting.

These figures are not directly applicable to line shafting in factories and the like for such shafting have the power applied as a rule, some place near the center of the shaft and the power is taken off at intervals along the shaft, whereas the preceeding assumes that the total power is conveyed from one end of the shaft to the other.

Shafts are usually made of cold rolled steel or iron or else from turned steel or iron. The cold rolled shafting comes from the rolling mills clean and polished with a uniformly circular cross section and very straight The cold rolling process compresses the metal on the outside and thereby raises the elastic strength. If

TABLE XI

LIMIT OF LENGTH FOR STEEL SHAFT

NO PULLEYS ON LINE

DIAMETER OF SHAFT IN INCHES	LENGTH IN FEET WHEN TOTAL POWER IS ABSORBED	LENGTH IN FEET WHEN EFFICIENCY IS 50 PER CENT	LENGTH IN FEET WHEN EFFICIENCY IS 75 PER. CENT
1	1 585	792	396
2	3 170	1 585	792
3	4 755	2 377	1 188
4	6 340	3 170	1 585
5	7 925	3 962	1 981

See page 34

TABLE XII

LIMIT OF LENGTH FOR WROUGHT IRON SHAFT.

NO PULLEYS ON LINE

DIAMETER OF SHAFT IN INCHES	LENGTH IN FEET WHEN TOTAL POWER IS ABSORBED	LENGTH IN FEET WHEN EFFICIENCY IS 50 PER.CENT	LENGTH IN FEET WHEN EFFICIENCY IS 75 PER.CENT.
1	1 250	625	312
2	2 500	1 250	625
3	3 750	1 875	937
4	5 000	2 500	1 250
5	6 250	3 125	1 562

See page. 34

key seats are cut in cold rolled shafting it is likely to spring the shaft as part of the compressed metal is removed from the out side and thus the strain set up causes the shaft to bend out of shape.

A shaft is usually subject to two kinds of stresses, namely a tortional stress and a transverse stress which causes a bending action. When the weight of the pulleys, couplings and the pull of the belts is small and the bearings are close together the bending stress is small as compared with the torsional stress and the bending action may be neglected

To determine the diameter of a shaft to transmit a given horsepower let P = force tending to turn the shaft

a = distance from axis of shaft at which P acts, in inches

T = Pa, = twisting moment,

J = Polar moment of inertia = $\frac{\pi d^4}{32}$ (20)

d = diameter of shaft in inches,

S = unit shearing resistance,

C = distance from axis to most remote fibre of the shaft = $\frac{d}{2}$

N = number of revolutions made by shaft,

HP = horse power transmitted

Since the resisting moment is equal to the twisting moment,

$$T = \frac{SJ}{C} = Pa$$

$$Pa = \frac{\frac{\pi d^4 S}{32}}{\frac{d}{2}} = \frac{\pi d^3 S}{16} = \frac{d^3 S}{5.1} = 0.1963\, d^3 S$$

$$\therefore d = \sqrt[3]{\frac{5.1\, Pa}{S}} \qquad (5)$$

20. See Merriman's Mechanics of Materials. pp 173.

Now the horsepower transmitted is

$$HP = \frac{2\pi N\ Pa}{12 \times 33000} = \frac{2\pi N\ T}{12 \times 33000}$$

$$\therefore T = Pa = 63025 \frac{HP}{N}$$

Substituting this value of T in equation (5) we get

$$d = \sqrt[3]{\frac{321,427.5}{S} \times \frac{HP}{N}} \qquad (6)$$

Thus far there have been no assumptions made but here we have to determine the unit shearing resistance of the metal which is being used and assume a factor of safety. This is what gives the many different formula for the diameter of shafts, which are to be found in the different books on the subject.

Using the allowable shearing stress of 64,000 pounds for cold rolled iron shafting and 33,500 for turned iron shafting and the factors of safety given in the table XIII the constants come out the same as those given by Jones & Laughlins.

Francis gives Table XIV for the greatest admissible distance between the bearings of continuous shafts, subject to no transverse strain except their own weight.

It will be seen from Table XIV that there is very little difference between the distance for wrought iron and steel and the distinction between the two is fast disappearing in this kind of work.

There is a great difference of opinion as to the speed at which shafts should be run but the general practice is to run line shafting in machine shops at from 85 to 180 R.P.M. while in wood working shops they are run from 125 to 300 R.P.M. Richards recommends the following speeds of line shafts for the different size shafting in machine and wood shops:

TABLE XIII

FORMULAE FOR DIAMETER AND HORSE-POWER OF COLD-ROLLED AND TURNED IRON SHAFTS.

DESCRIPTION OF SHAFT	FACTOR OF SAFETY	MATERIAL			
		COLD ROLLED UNIT STRESS 64,000#		TURNED IRON UNIT STRESS 33,500#	
	f	d	H.P.	d	H.P.
FOR HEAD SHAFTS SUPPORTED BY BEARINGS CLOSE TO EACH SIDE OF THE MAIN PULLEY	15	$\sqrt[3]{\frac{100\ H.P.}{N}}$	$\frac{Nd^3}{100}$	$\sqrt[3]{\frac{125\ H.P.}{N}}$	$\frac{Nd^3}{125}$
FOR LINE SHAFTING FROM WHICH POWER IS TAKEN AT INTERVALS	10	$\sqrt[3]{\frac{50\ H.P.}{N}}$	$\frac{ND^3}{50}$	$\sqrt[3]{\frac{90\ HP}{N}}$	$\frac{Nd^3}{90}$
FOR SIMPLY TRANSMITTING POWER AND SHORT COUNTERS	6	$\sqrt[3]{\frac{30\ H.P.}{N}}$	$\frac{Nd^3}{30}$	$\sqrt[3]{\frac{50\ HP}{N}}$	$\frac{Nd^3}{50}$

H.P. = HORSE POWER ; d = DIAMETER OF SHAFT ; N = REV. PER MIN.

See page 38

TABLE XIV

GREATEST ADMISSABLE DISTANCE BETWEEN SHAFT BEARINGS.

DIAMETER OF SHAFT IN INCHES	DISTANCE BETWEEN BEARINGS IN FEET.	
	WROUGHT IRON	STEEL
2	15.46	15.89
3	17.70	18.19
4	19.48	20.02
5	20.99	21.57
6	22.30	22.92
7	23.48	24.13
8	24.55	25.23
9	25.53	26.24

Table XV.

Speed of Line Shafts.

Diameter of shaft in inches	1½	1¾	2	2¼	2½	2¾	3	3½	4
R.P.M. for wood working machinery	300	275	250	225	200	180	170	160	150
R.P.M for metal working machinery	150	140	130	120	110	100	95	90	85
R.P.M. for long shafts conveying power between two points	300	280	260	240	220	200	180	160	140

It is best practice to have one diameter throughout for each line of shafting and also to have the hearings spaced equally. Then the parts are all interchangeable. It is best to have shafting of two inches or more in diameter so arranged that there is one coupling and one hearing to each section The first or starting section is often made larger which is not objectionable and should be made larger than the other sections so that it will receive two bearings in order to make it more easily erected.

Richards gives the following table for the lengths of the sections of shafts thus arranged:

Table XVI

Length of line shafts, Short sections

Diameter in inches	2	2¼	2½	2 3/4	3	3½	4
First section, feet	10	11¼	12½	13 3/4	15	17½	20
Other sections, "	8	9	10	11	12	14	16

Couplings.

In order to connect the sections of shafting couplings are used There is a great variety of these in use. Couplings may be classiified in the following manner:

(1) Permanent or fixed couplings

(a) Box or muff couplings

(b) Clamp or compression couplings

(c) Double cone vice coupling

(d) Flange coupling.

(2) Loose or disengaging couplings

(a) Claw couplings or clutch

(b) Friction coupling or clutch

Permanent or fixed couplings are those which can be disconnected only by unscrewing bolts or slacking keys. These couplings are used to connect two sections of line shafting which are run together and do not require to be disconnected. The loose or disengaging couplings are used between two sections of paraldel shafting one of which is run only for part of the time.

The box or muff coupling is not often used in this country, but in England it is used to some extent in the cotton mills. This coupling consists of a wrought or cast iron cylinder the inside ends of which are bored out to fit the shafts, with a key way cut on the inside. There is also a key-way cut on the ends of the shaft and when the collar is placed on the ends of the shafts and the key driven in place there is a firm, rigid coupling. This form of coupling is very expensive as it is not interchangeable and must be fitted to each shaft on which it is used. The advantage of it ,though, is that it forms a firm union between the shafts and presents no points upon which clothing can be cought. Figure 11 shows this form of coupling.

The clamp or compression coupling is the kind most used in

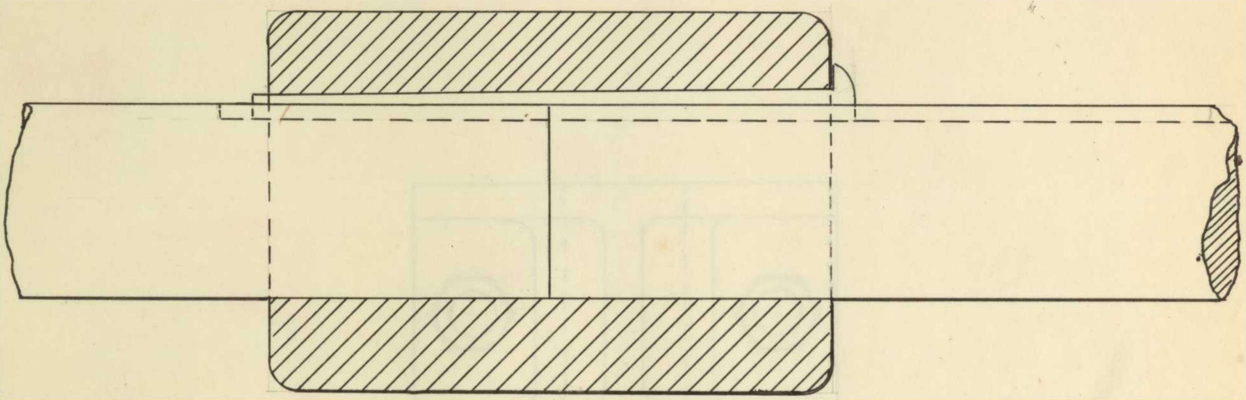


Figure 11.

machine shops and factories in this locality. This coupling consists of a hollow cylinder split longitudinally with flanges along either side of the division which are drilled so that the halves may be bolted together. There is a key-way cut on the inside of the coupling. Figure 12 shows the usual form of this coupling A shell is often screwed to the outside of the coupling which makes it safer as the bolts are protected and there are no projections to catch the clothing of workmen. When the shell is fitted to the coupling a belt may run on it. This coupling has the advantage over the box coupling in that it may be put on or taken off without moving the shafting.

The double-cone vice coupling was introduced by Mr Sellers of the Wm. Sellers & Co. This coupling is shown in Figure 13. It consists of an outer cylindrical barrel enclosing the ends of the sh shafts. The inside of this is turned to a double conical form Be-

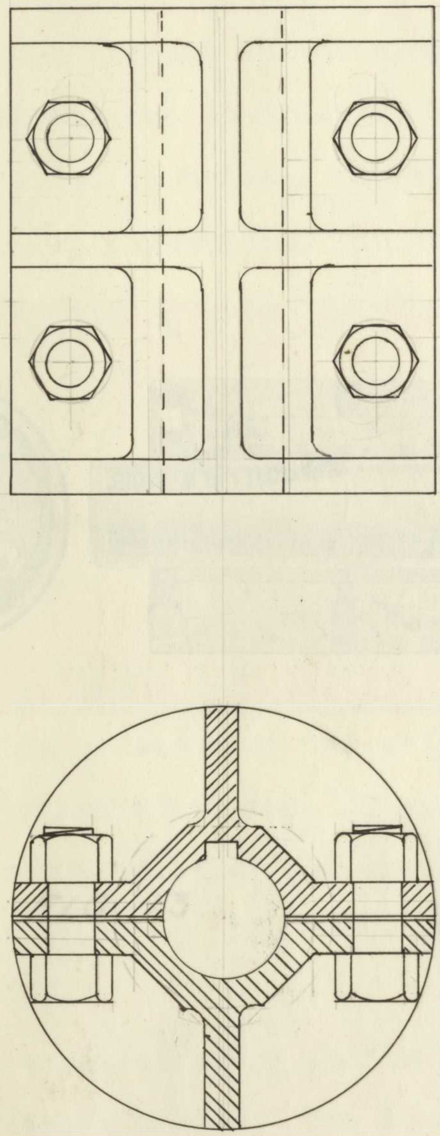


FIGURE 12.

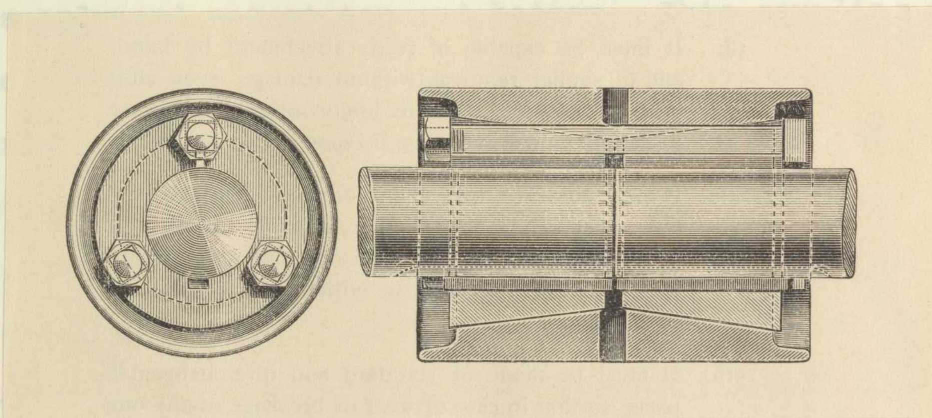


FIG. 13.

tween the barrel and the shaft are two sleeves the outside of which are conical, and fit the box and the inside cylindrical and fit the shaft. These sleeves are pressed together by three screw-bolts parallel to the shaft. The bolts are square in section and rest in slots cut into the sleeves and the barrel. To give elasticity to the sleeves, they are cut through on one side, at the bottom of one of the bolt slots. Each sleeve is drawn in with equal force and grasps the shaft with equal tightness. A key is driven into each shaft end as an additional precaution, but these keys should fit sideways only and not at top and bottom. This coupling can be used to connect two different sized shafts In this case a coupling of the standard size of the larger sized shaft is used with one of the sleeves bored for the smaller sized shaft. The advantage of this coupling over the two previous couplings is that two sizes of shafts may be coupled without turning the larger shaft down to the size of the smaller which is necessary when using the other couplings.

The flange coupling consists of two cylindrical barrels at one end of which there is a radial flange. These flanges have holes drilled in them to match each other and are bolted together. The barrels are keyed to the shaft. Figure 14 shows this form of coupling.

This coupling is employed where shafts have to be temporarily disconnected, and for heavy shafts. The bolts connecting the flanges must be heavy enough to resist the shearing stresses and must be a tight fit.

Unwin[21] gives the following formula for the number of bolts:

$$N = 3 + \frac{D}{2}$$

---

21. Machine Design, Unwin, Vol. I, pp 234.

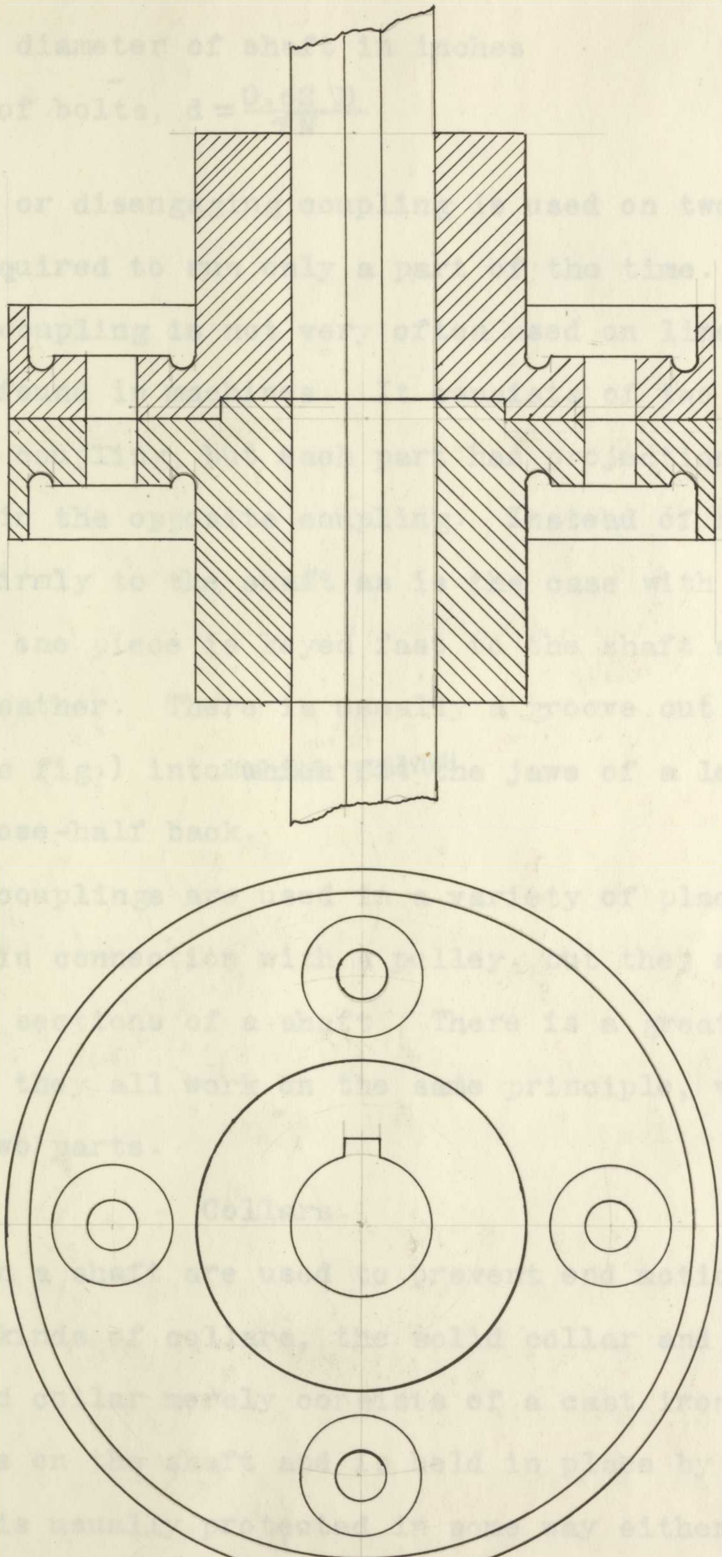


FIGURE 14.

where D is the diameter of shaft in inches

Diameter of bolts, $d = \frac{0.62\ D}{\sqrt{N}}$

The loose or disengaging coupling is used on two shafts one of of which is required to run only a part of the time.

The claw coupling is not very often used on line shafts, but is more often found in machines. It consists of two parts, like the face plate coupling, but each part has projections which fit into recesses in the opposite coupling. Instead of having both halves keyed firmly to the shaft as is the case with the flange coupling, only one piece is keyed fast to the shaft and the other works upon a feather. There is usually a groove cut around the loose half (see fig.) into which fit the jaws of a lever for sliding the loose-half back.

Friction couplings are used in a variety of places. Most of them are used in connection with a pulley, but they are often used to connect two sections of a shaft. There is a great variety of them; however, they all work on the same principle, vis, the friction between two parts.

Collars.

Collars on a shaft are used to prevent end motion of the shaft. There are two kinds of collars, the solid collar and the split collar. The solid collar merely consists of a cast iron or forged ring which fits on the shaft and is held in place by a set screw. The set screw is usually protected in some way either by flanges on the ends of the collar or else the screw has no head and is flush with the outside of the collar and is tightened with a screw driver. Figure 15 shows this kind of collar.

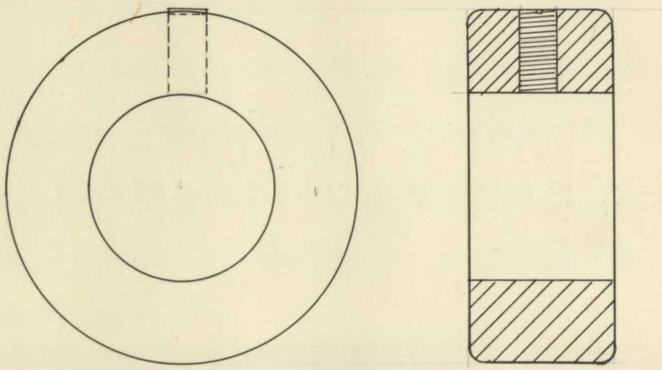


Figure 15.

The split collar is just like a clamp coupling only that it is much shorter. Figure show several forms of split collars The split collar has the advantage over the solid collar that it can be put on or taken off a shaft without taking the shaft down.

Bearings.

The economical transmission of power by shafting depends more upon the bearings than any other one thing. The loss in power due to friction of shafting in its bearings sometimes is as high as 60% of the entire power. Flather finds from the results of tests made upon well known factories that the power required to drive the shafting ranges from 15% to 80% of the total power.[22]

Professor C. H. Benjamin finds from tests made in sixteen man-

---

22. Dynamometers and the Measurment of Power. J.J.Flather.p5.Trans-

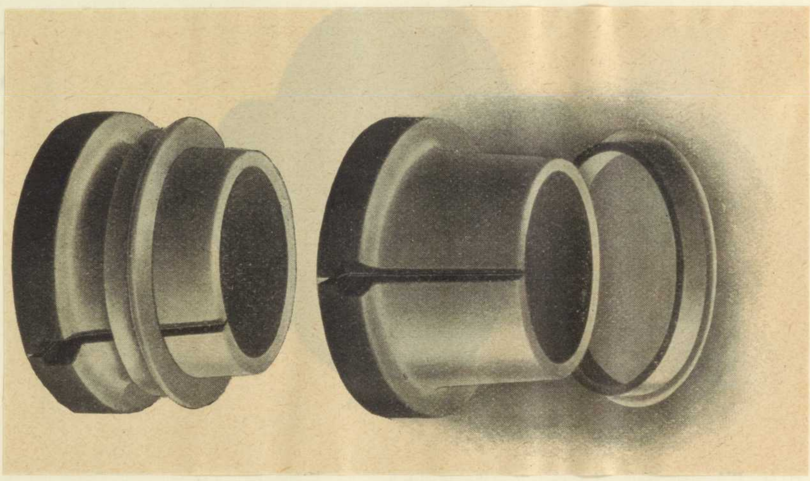


FIGURE 16.

ufacturing establishments that the maximum amount of power to drive shafting was 80.7% of the entire power and the minimum was 14.5% of the entire load.[23] The shafting in the factory using the minimum amount of power to drive the shafting runs in castiron bearings and is oiled by hand instead of by wick oilers.

The bearings should be pivoted so as to allow of longer bearings and a corresponding reduction of pressure on their surfaces. If the shafting were perfectly straight, pivoted bearings would not be necessary. The length of the bearing should be about four times the diameter of the shaft.

---

23. Transactions of American Society of Mechanical Engineers. Volume XVIII, pp 288 - 235.

UNIVERSITY OF ILLINOIS WOOD-SHOP INSTALLATION.

Plate I shows the installation of power in the new wood-shop building of the University of Illinois.

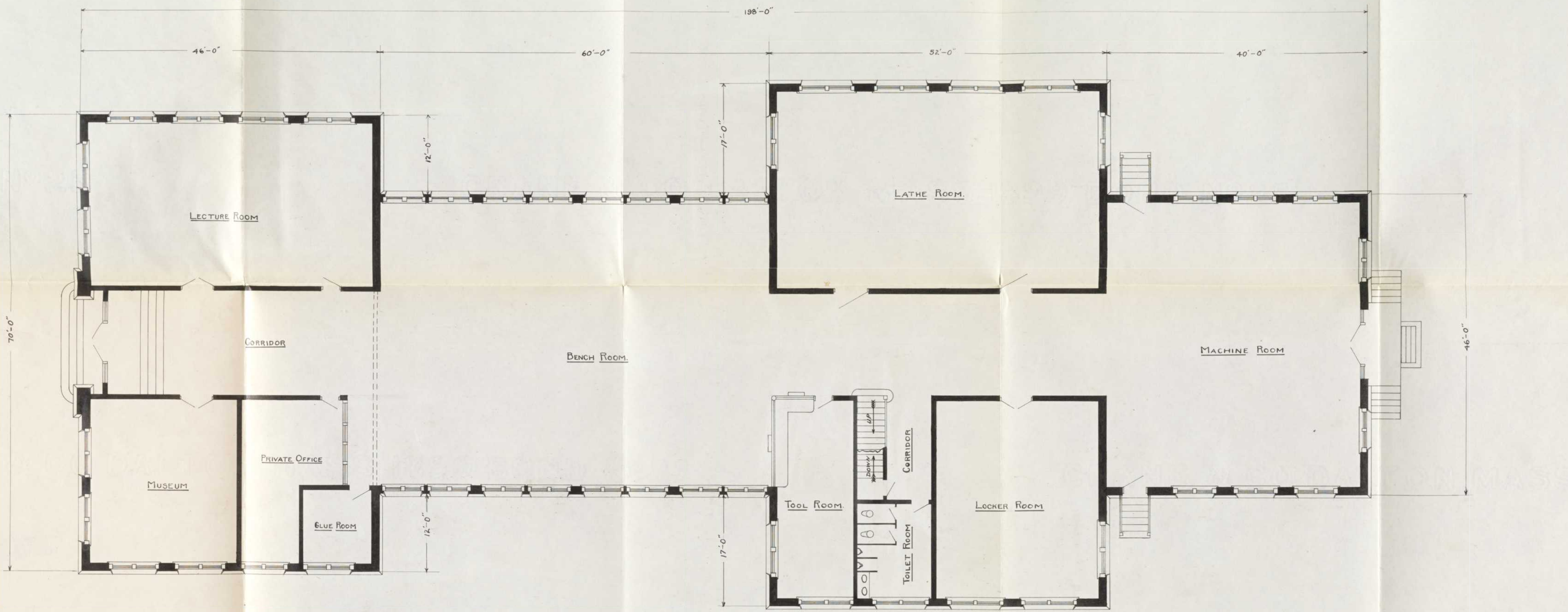
All the machinery is on the one floor and the shafting is placed beneath the floor wherever it is possible. In the machine room where all the machinery is except the lathes, there is no shafting or belting in sight. In the lathe room the shafting is overhead All visible shafting and belting is shown in full in the drawing and the shafting and belting under the floor is shown in dotted lines

The main line shaft which is marked A is driven by a 20-horse-power polyphase induction motor which is marked 1 on the drawing. The shaft is connected to the motor by an 8" double leather belt with a 9" pulley on the motor and a 40" pulley on the main shaft. The machines in the machine room are all belted to their counter shafts beneath the floor and they to the main shaft.

Shaft A drives shaft B by means of two 24" pulleys and a 6" double leather belt. This shaft only acts as a counter shaft for the lathe room. There is a friction clutch pulley on the shaft B carrying a 6" double belt which drives the shaft C. These three shafts are belted so as to run at the same speed of 300 revolutions per minute. Shaft C is belted to D by a 4" double leather belt and C is belted to e by a 5" double belt. D and E run at the same speed as C.

Shafts A, B and C are 2 3/8 inches in diameter and are sufficient ly large to transmit the power. Shaft D, and E are 2 inches in diameter and are likewise sufficiently large .

The belts have likewise been found sufficiently large to transmit the power.



PLAN
OF
THE WOOD SHOP BUILDING
THESIS DRAWING

Mech. Eng'g. Dep't.

Scale $\frac{1}{8}$" = 1'. University of Illinois

PLATE II Harry McCarthy JUNE 1902